RESEARCH ARTICLE

Experimental Investigation of Natural Frequency of a Cantilever Composite Beam using Vibration Absorber

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ABSTRACT: Composite materials are used in utmost all aspects of the industry and commercial fields such as in aircraft, ships, common vehicles, etc. The most attractive properties of such materials include high strength-to-weight ratio and high stiffness-to weight ratio. In this paper, a composite cantilever beam specimen of size 300 x 25 x 2.5 (mm) is prepared by natural fiber material (sisal) and it is used as a basic prototype for a number of complex flexible engineering structures which is highly resonant in characteristic. If the amplitude of vibration of a cantilever beam becomes excessively large, ultimately the system tends to fail. To eliminate such vibrations, a spring mass system is attached with vibration absorber to control the structural resonance. This problem is solved by Finite Element Method using ANSYS. Then the FEM results are verified by conducting experiment. Using FEM, the excitation is given to the beam the amplitude response without absorber is found by harmonic analysis. Then the absorber is attached at different locations on the beam, the absorber stiffness is varied until the amplitude of vibration in the beam is reduced to minimum. Then the beam is excited by the electro-dynamics shaker and the amount of vibration is measured using accelerometer. The output from the accelerometer is in analog form which is then converted into digital signal by means of data acquisition system. The circuits are constructed by Lab VIEW software to convert the digital signal into amplitude of vibration. Finally the experimental results were compared with FEM results for validation. Experimental results are found to be close agreement with FEM results.

KEYWORDS: Amplitude, Composite beam, Natural Frequency, FEM and Vibration absorber

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I.INTRODUCTION

Early researchers in field of vibration intense their efforts on understanding the natural phenomena and developing mathematical theories to describe the vibration of a physical system [1]. In recent times, many investigations have been motivated by the engineering applications of design, such as design of machines, foundations, structures, engines, turbines and control systems [2,3].Vibration in most of the engineering applications is undesirable and has to be eliminated, this is not practically possible in most of the cases, but it has to be reduced to as minimum as possible [4]. Excessive vibration causes discomfort to human beings, damage to buildings and rapid wear of machine parts such as bearings and gears. At resonance it may even lead to shut down of the turbine units. Resonance is the stage where the natural frequency of the system is equal to the external excitation frequency [5-7]. At resonance the amplitude of vibration is excessive. So, determination of the natural frequency of the system is a must from the design point of view. If an external energy source is applied to initiate vibrations and then removed, the resulting vibrations are said to be free vibrations which is taken into consideration. When a structure externally excited has undesirable vibrations, it becomes essential to reduce them by coupling some vibrating system to it. The vibrating system is known as vibration absorber or dynamic vibration absorber. In such cases the excitation frequency is nearly equal to the natural frequency of the structure or machine. The mass (machine or structure) which is excited can have zero amplitude of vibration and the spring mass system (absorber) which is coupled to it vibrates freely [8,9].The dynamic vibration absorber has been effectively used for controlling the vibration in a large number of practical situations. For example, it has been applied in various forms to reduce the Vibration of the body of a small electric hair clipper, Torsional oscillation of the crankshaft of an engine, Rolling motion of a ship or yacht, Vibration of long overhead transmission lines, Chatter of a cutting tool, Noise in an aircraft cabin etc.

The dynamic vibration absorber is the mass–spring system, which is attached to the main system. The natural frequency of the attached absorber is chosen to be equal to the frequency of the disturbing force [8]. It will be shown theoretically that the main mass does not vibrate at all and that absorber system vibrates in such a way that their springs force is at all instants equal and opposite to forcing function. When tuned appropriately this arrangement was theoretically capable of setting the main system vibration to zero at a particular frequency [9]. An experimental investigation of a particle damping method for a beam and a plate was proposed [10-12]. Tungsten carbide particles are rooted in longitudinal (and latitudinal) holes drilled in the structure, as a simple and passive means for vibration Suppression. Mechanisms of energy dissipation of particle damping are highly nonlinear and mainly linked to friction and impact phenomena. Experiments are conducted with a number of arrangements of the filled particles including different element sizes and volumetric packing ratios [13,

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14]. The particle damping is extremely efficient and that strong attenuations are achieved within a broad frequency range. A Magnetic damping method is used to attenuate a vibration of a beam in which the

magnetic fields may be used to apply damping to a vibrating arrangement. Dampers of this type function during the eddy currents that are generated in a conductive material experiencing a time changing magnetic field [15, 16]. The density of these current is directly related to the velocity of the change in magnetic field. Due to the generation of these currents, the internal resistance of the conductor causes them to drive away into heat. Since a portion of the moving conductor's kinetic energy is used to generate the eddy currents, which are then dissolute, a damping achieve occurs. An adaptive resonant controller is used to attenuate vibrations in a beam of cantilever structure with large varying parameters [2, 17]. This controller is particularly suited for structures that are exposed to previously un-modeled dynamics. The structures current natural frequency is estimated by online identification of its Eigen values. The performance of the proposed adaptive resonant controller is compared with that of a fixed parameter resonant controller. Initially the modal analysis in ANSYS is carried out to find the beam's natural frequency and to form the transfer function of the beam. Zero crossing method is used to make a resonant controller adaptive to measure the system's frequency. Thus for multiple-frequency excitations, a parameter identification method will need to be used in place of the zero-crossing method then the adaptive resonant controller is made and the frequency response for each model is determined. An adaptive resonant controller is used to attenuate multimode vibrations is a flexible beam of cantilever structure with varying load conditions [17]. The adaptive scheme is realized by on-line estimation of the system's natural frequencies. The multimode estimator is implemented using a set of parallel second-order recursive least square estimators. A study also proposed by [7] in which numerous small spring-mass-damper systems are attached to a huge hanging mass instead of the master structure. The harmonic and impulse response of two structures with numerous attachments, one with a discrete number and other with a continuous master structure, were compared to investigate the effects of modal overlap and dissipation In discrete model, is a single degree of freedom master structure with numerous small spring mass damper system attached to it. The attached systems form the structure that acts as a vibration absorber tuned over a frequency band centered about the master structure's isolated natural frequency. The continuous model has the same substructure as a discrete model, but the master structure is a cantilever beam. Analytical and Numerical Combined Method (ANCM) was determined in which the natural frequencies and corresponding mode shapes of a uniform beam of cantilever carrying many number of elastically point mass mounted [18]. This method found to be much better than the conventional Finite Element Method [19, 20]. So many researches has done in the field of composite fiber based work, whenever a new fiber is introduced as composite materials, the properties checking are not only the best option to predict the material quality. This work aimed to find the frequency of a composite strip under cantilever condition with various damping force.

Form the above survey, the spring-mass system has not been used as an absorber to reduce the amplitude of vibration in the composite cantilever beam. Due to this reason spring-mass system is preferred as an absorber to reduce amplitude of vibration in a beam which is excited by a shaker. Mostly it dealt with other types of damping methods, natural frequencies and mode shapes of a beam.

II. EXPERIMENTATION

A. Theoretical Formulation

System excitation is a sinusoidal forcing function of amplitude F and frequency ω applied to the primary mass. If the excitation frequency ω is nearly close to the natural frequency

$\omega = \omega_n = \sqrt{k_1}/m_1$

of the system, the amplitude of vibration would be very large because of resonance. Fig. 1. shows single degree of freedom and after coupling the spring mass system, it becomes two degrees of freedom system.

The equations of motion can be written as

$$m_1x_1 + k_1x_1 + k_2(x_1 - x_2) = F\ Sin\omega t \quad (1)$$

$$m_2x_2 + k_2(x_2 - x_2) = 0 \quad (2)$$

Let us assume the solution of the form

$$x_1 = A_1 sin\omega t \ and\ x_2 = A_2 sin\omega t \quad (3)$$

Substituting the values and solving the above equations

$$A_1 = \frac{(m_2 - m_2\omega^4)\,F}{\beta} \quad (4)$$

$$A_2 = \frac{(k_2)\,F}{\beta} \quad (5)$$

Where

$$\beta = m_1m_2\omega^4 - (m_1k_2 + m_2(k_1 + k_2))\omega^2 + k_1k_2$$

In order to have the amplitude of mass m_1 as zero, let us consider the equation

$$A_1 = \frac{(m_2 - m_2\omega^4)\,F}{\beta} = 0$$

$$\omega = \omega_2 = \sqrt{k_2}/m_2$$

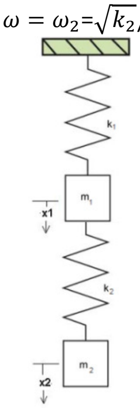


Fig. 1. Spring mass system

Thus the mass and spring constant of the absorber system are selected in such a way that the above equation is satisfied it becomes a dynamic absorber system. In this paper, the cantilever beam is considered as a primary system.

TABLE I List of Nomenclatures

Symbol	Quantity	SI Units
ω_n	Natural frequency	Hz
βn	Frequency Constant	-
l	Length of beam	mm
E	young's modulus	N/mm^2
A	Amplitude	N/mm^2
I	Moment of inertia	mm^4
ρ	Density of the material	kg/mm^3
x	Distance	mm
k	Stiffness of spring	N/mm
m_1	Mass of primary system	kg
m_2	Mass of absorber system	kg
k_1	Stiffness of primary system	N/mm
k_2	Stiffness of absorber system	N/mm
ω	Frequency of forcing function	Hz

The natural frequency is calculated theoretically for the beam structure using the relation

$$\omega_n = \frac{(\beta_n l)^2 \ \sqrt{EI}}{\rho A}$$

$$\omega_n = 133.15 \text{ rad /sec}$$

The maximum natural frequency found in theoretically is 133.15rad/s.

B. Experimental Setup

Sisal fiber is taken for making composite bar. A mild steel mould box is prepared as 300 mm x 25 mm per requirement in which raw material (Sisal) is adding with hardener (Resin) was mixed uniformly with 1:3 ratio and then heated at a temperature of 60° C and maintained 10 to 15 minutes by wet lay method after that another layer was added with same process and this was continued till 2.5 mm thick reached. The molded composite was kept over eight hours to the atmospheric conditions. The specimen of the composite has the dimension of 300 mm x 25 mm x 2.5 mm as shown in Fig.2. Table II shows the geometry and properties of the specimen.

Fig. 2.Specimen sisal composite material

TABLE II Geometric and Properties of Specimen

Quantity	Symbol	Unit	Description
Raw material (fibre)	-	-	Sisal
Length	l	mm	300
Breadth	b	mm	25
Thickness	t	mm	2.5
Area	A	m^2	7500
Moment of Inertia	I	mm^4	2083.3
Young's modulus	E	N/m^2	12×10^9
Density	ρ	kg/m^3	1450
Poisson's ratio	γ	-	0.2
Resin Binders		-	Resins and Hardeners
Resin Mixing ratio	-	-	1: 3

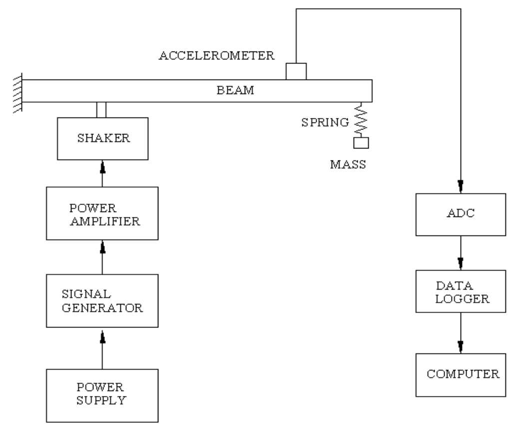


Fig. 3.Schematic experiment layout

The composite beam was fixed as a cantilever beam. The electro dynamic shaker is used to excite the cantilever beam. The accelerometer is attached with the beam is used to measure the vibration of a beam. Then the output of accelerometer is given to the electronic circuit which converts the analog output of accelerometer into digital output. The schematic experimental layout is shown in Fig. 3. The vibration exciter is already connected to the power amplifier. The function /signal generator is connected to both channel of a frequency meter to measure the excitation frequently and the power amplifier. The composite specimen is placed into the vibration dynamic shaker. The amplitude of the vibration was recorded in three different location of the cantilever composite beam with the distance at 300 mm from the fixed end (free end), 250 mm and 200 mm from the fixed end.

Fig. 4. Shows the experimental setup with data logging system for the experimentation. The frequency recorded on the cantilever beam was measured with the help of frequency analyzer. Once first node was noticed, start using the fine adjustment knob in the frequency meter. After measuring the frequency at which the first vibration occurs, continue this to increase the frequency until to reach the second and third modes and so on. This measuring procedure is repeated for each mode. After the turn off the power amplifier and adjust the composite specimen to a different length and tighten. The computer technology is used for getting amplitude of vibration and graphs. Finally switch the specimen to the composite specimen records length, and measuring three nodes like 300mm, 250mm,

200mm. Finally the optimum values of absorber stiffness, mass and location of the absorber is found for which the amplitude of vibration is analyzed. The excitation point is changed, the above procedure is carried out and the results are tabulated.

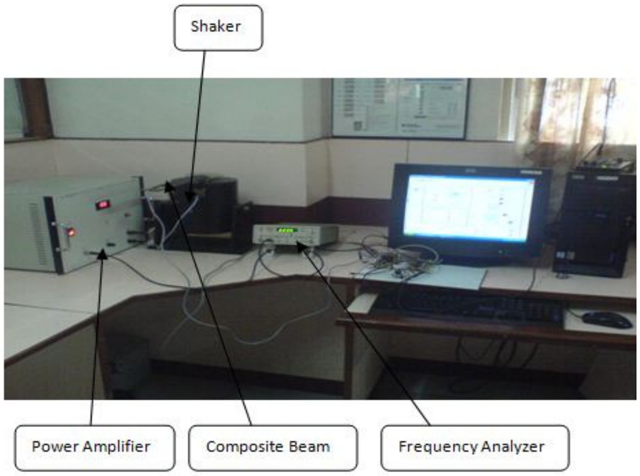


Fig. 4.Experiment setup with data logging system

C. Case Study

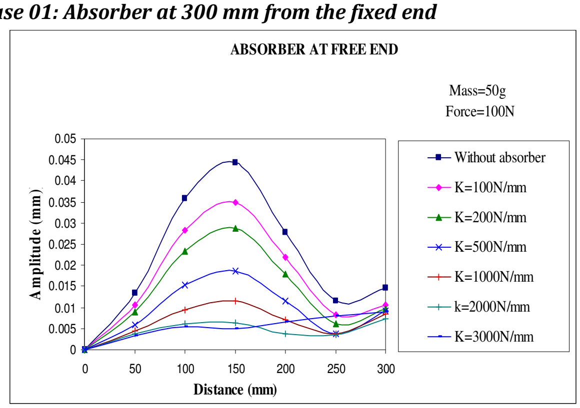
The experiment was carried out with following three cases and the results were compared with and without using absorber in the composite cantilever beam. Fig. 5. Shows the experimental setup of; a) without absorber, b) Absorber at 300 mm from the fixed end, c) Absorber at 250 mm from the fixed end, and d) Absorber at 200 mm from the fixed end.

Case 01: Absorber at 300 mm from the fixed end
Case 02: Absorber at 250 mm from the fixed end
Case 03: Absorber at 200 mm from the fixed end

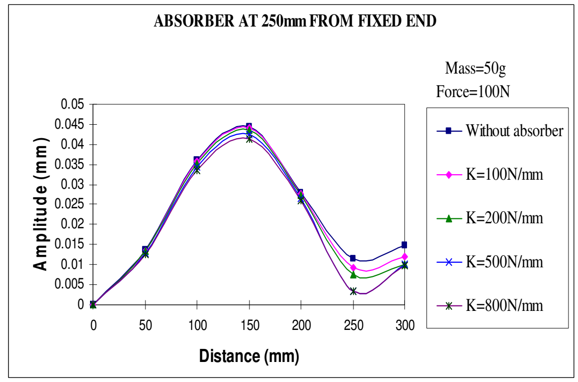
III. RESULTS AND DISCUSSION

Distance vs. Amplitude

Case 01: Absorber at 300 mm from the fixed end



Case 02: Absorber at 250 mm from the fixed end



Case 03: Absorber at 200 mm from the fixed end

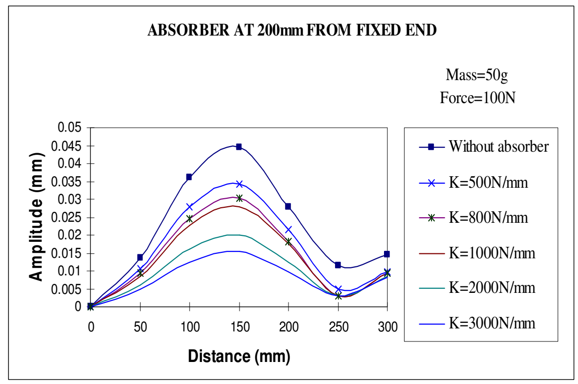


Fig. 6. Distance vs. Amplitude Case 01 - Absorber at 300 mm from the fixed end, Case 02 - Absorber at 250 mm from the fixed end, and Case 03 - Absorber at 200 mm from the fixed end.

In the case 01, the maximum amplitude of the beam without absorber is found to be 0.045 mm at the distance of 150 mm from the fixed end. Various range of stiffness value such as 100 N/mm, 200 N/mm, 500 N/mm, 1000 N/mm, 2000 N/mm, and 3000 N/mm are consider for amplitude. At the stiffness k = 3000 N/mm, the amplitude is gradually increased as shown in figure. From the observation it is noted that upto 2000 N/mm stiffness the amplitude is decreased, while the case of stiffness k = 3000 N/mm the amplitude is gradually increased at the distance of 150 mm from the fixed end. This is the reason why k = 2000 N/mm is consider for the above (all) conditions. Thus the maximum amplitude of the beam with and without using the absorber is found to be 0.015 and 0.005 respectively.

To reduce the vibration, the mass of 50 grams with cyclic load of 100 N was suspended at the free end of the beam (i.e. 300 mm). After then the amplitude is reduced from 0.045 mm to 0.005 mm as shown in the fig. The graph is plotted between distance versus amplitude from the fixed end of 300 mm with and without using absorber by considering the stiffness k =2000 N/mm.

The above procedure has been repeated for case 02 (when the absorber is at 250 mm from the fixed end) and case 03 (when the absorber is at 200 mm from the fixed end), for the condition of the beam with and without using the absorber. The results are tabulated in the table III & IV and the graphs are plotted in the Fig. 6 & 7.

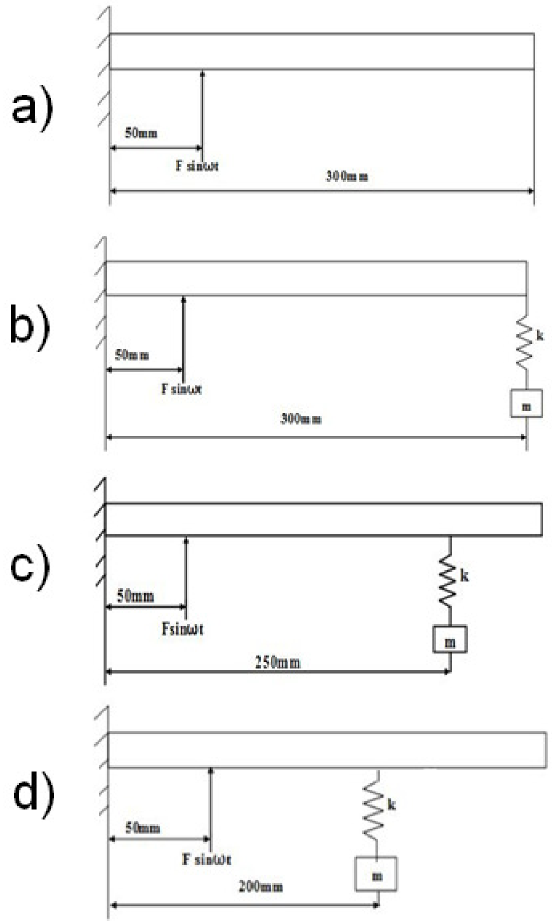


Fig. 5. Experiment setup a) without absorber, b) Absorber at 300 mm from the fixed end, c) Absorber at 250 mm from the fixed end, and d) Absorber at 200 mm from the fixed end.

TABLE III Experimental results Excitation at 50 mm from Fixed End

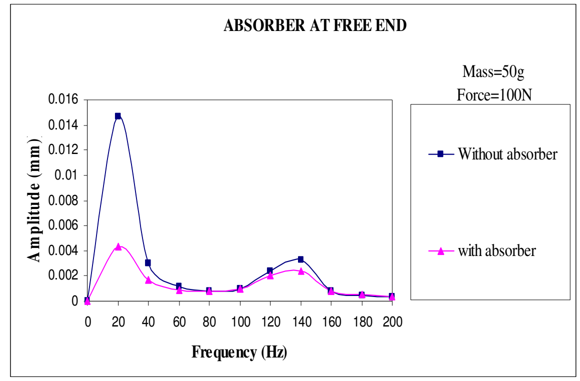
Absorber mass (gram)	Absorber Stiffness (N/mm)	Maximum amplitude without absorber (mm)	Position of absorber from fixed end (mm)	Maximum amplitude with absorber (mm)	% reduction in amplitude
50	2000	0.045	300	0.005	88
50	800	0.045	250	0.04	11
50	3000	0.045	200	0.015	66

TABLE IV Experimental results w.r.t position of the absorber from theend

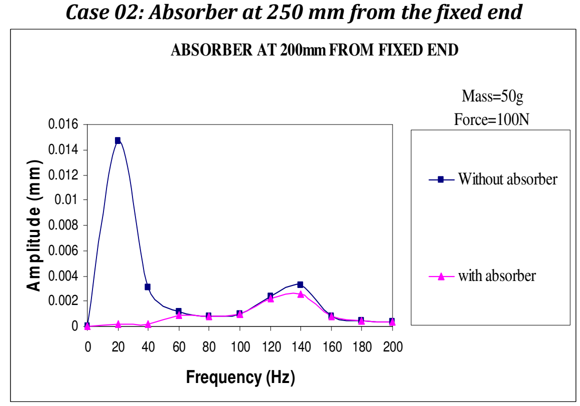
Absorber mass (gram)	Absorber Stiffness w.r.t beam stiffness (N/mm)	Position of absorber from fixed end (mm)	Maximum amplitude without absorber (mm)	Maximum amplitude with absorber (mm)	% reduction in amplitude
50	1.3	300	0.045	0.005	76
50	0.55	250	0.045	0.040	11
50	2.07	200	0.045	0.015	56

Frequency vs. Amplitude

Case 01: Absorber at 300 mm from the fixed end



Case 02: Absorber at 250 mm from the fixed end



Case 03: Absorber at 200 mm from the fixed end

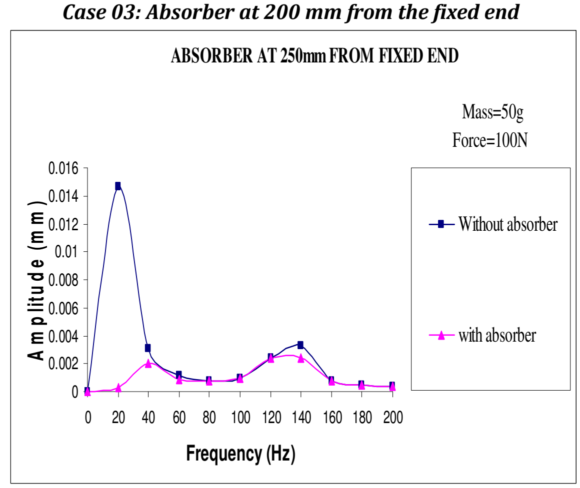


Fig. 7. Frequency vs. Amplitude Case 01 - Absorber at 300 mm from the fixed end, Case 02 - Absorber at 250 mm from the fixed end, and Case 03 - Absorber at 200 mm from the fixed end.

FEM Analysis

Finite element analysis has been made for the excitation at 50 mm from the fixed end with and without absorber is shown in Fig. 8. Table V shows the comparison between FEM and Experimental results.

Comparison of Experimental and FEM

Fig. 9, shows that the distance versus amplitude for the experiment conducted with and without absorber. It is noted that without absorber the amplitude of the specimen was found as 0.045 mm and when using absorber at the end the value of amplitude is found to be 0.005 mm. From this the

amplitude of vibration is found to be minimum while using the absorber.

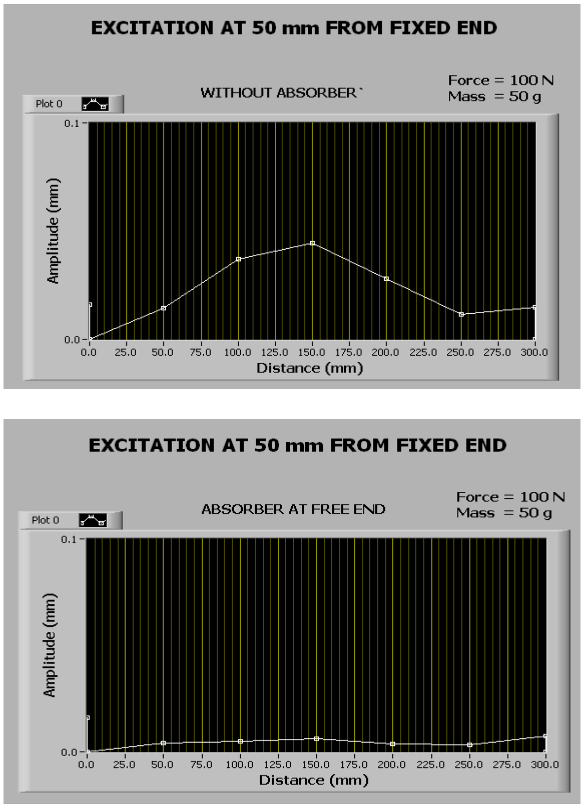


Fig. 8. FEM analysis on Distance vs. Amplitude with and without absorber

Fig. 9. Distance vs. Amplitude with and without absorber

TABLE V Comparison between Fem and Experimental results

Method of Study	Excitation from fixed end (mm)	Maximum amplitude without absorber (mm)	Maximum amplitude with absorber (mm)	% reduction in amplitude
FEM	50	0.051	0.006	87
Experimental Study	50	0.045	0.005	88

Table V shows the comparison of FEM and experimental results for the excitation from the fixed end at 50 mm. The reduction of amplitude in FEM and Experimental results are 87 % and 88 % respectively in which the results are closer.

CONCLUSION

In general, composite materials attracting more interest on wide range of engineering applications such as ship, aircraft, automotives, etc. In this study, sisal fiber composite material was prepared and the physical properties were tested. The specimen composite was taken for natural frequency experiment setup with shaker and data logging arrangements with and without absorber. From the experimental results, it is noted that the amplitude of vibration was decreases with increase in stiffness. The amplitude of vibration without absorber is very high and when the absorber is added at the end of the specimen (300mm from the fixed end) the amplitude was decreased. At the same time the absorber is placed at 250 mm and 200 mm from the fixed end, the amplitude of vibration was reduced further. The same has been validated with the FEM analysis without absorber. The comparison of the FEM and the experimental results has made, were 87% and 88% reduction in amplitude have observed with FEM and experimental results respectively. However the variation of the amplitude in the comparison was found due to the impact of environmental factors associated with the experimental results.

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